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A study of non-wetted heat transfer with water in teflon tubes

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A STUDY OF
NON-WETTED HEAT TRANSFER
WITH WATER IN TEFLON TUBES

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by

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(Date) Professor in Charge

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Edward J. Fritz

Edward J. Fritz

TABLE OF CONTENTS

ABSTRACT

Overall heat transfer coefficients are reported for the non-wetted flow of water in Teflon tubes over the range of Reynold's numbers from 3,000 to 50,000. Three Teflon tubes, of 3/8, 5/16, and 1/4 inch nominal inside diameter, were investigated using a one tube shell-and-tube heat exchanger with steam condensing outside the tube. Difficulty was experienced in obtaining consistant and reproducible data, probably owing to the elasticity of the Teflon tubes. The tube diameters are believed to change sufficiently with the pressure of the water in the tubes to cause erratic results.

The high wall resistance of the tubes owing to the low thermal conductivity of Teflon, limits the heat flux and masks changes in the heat transfer with water rate. The high wall resistance also leads to difficulty in isolating the water film coefficients by use of the Wilson plot.

The overall heat transfer coefficients for the tubes ranged from 43 to 74 BTU/hr-ft^2-°F.

INTRODUCTION

In recent years the use of liquid metals for heat transfer fluids has generated interest in the flow and heat transfer characteristics of non-wetted systems. The commonly used liquid metals, such as mercury and sodium, do not wet steel or copper tubes. This phenomena of non-wetting is believed to be a cause of much disagreement in liquid metal heat transfer data. This is possible because a small change in the composition of a liquid metal or its container will often change the wetting characteristics of the system appreciably.

Water, and most other fluids, do not wet Teflon* (Ref. 5). Heat transfer and fluid flow, accompanied by non-wetting, can be studied with water flow in Teflon tubes without the experimental difficulties associated with liquid metal heat transfer work. This does not mean that water flowing in Teflon tubes is completely analogous to liquid metal flow systems. The Prandtl numbers of liquid metals range around 0.1 while the Prandtl number of water is around 10 at room temperature. Since the Prandtl number represents the ratio of molecular diffusivity of momentum to molecular diffusivity of heat, the temperature gradients and mode of heat transfer in the laminar and buffer regions, where the greatest heat transfer resistance occurs, will be quite different for water and liquid metals.

* Teflon is the DuPont trade name for a polymer of tetrafloroethylene.

Tausch (Ref. 10) studied Fanning friction factors for flow of water in Teflon tubes. He found no significant difference in the flow characteristics of this non-wetted system from the friction factor data for wetted systems. Therefore according to the analogy between heat and momentum transfer there should be no difference in heat transfer rates for wetted and non-wetted systems. However an electrical resistance has been measured at non-wetted interfaces which is not present at wetted interfaces. This indicates there may also be an additional thermal resistance present, due to non-wetting, which is not accounted for in the heat and momentum transfer analogy.

The purpose of this investigation is to extend the work started at Lehigh several years ago (Ref. 1) by studying the inside film heat transfer coeffients of the Teflon-water system. It is desired to know if there are any significant differences in the size and variation of inside coefficients for non-wetted systems compared to conventional wetted systems which can be attributed to the non-wetting phenomena.

THEORETICAL BACKGROUND

The individual film coefficients for heat transfer from steam condensing on the outside of a tube to water flowing inside the tube are defined:

$$h_1 = q/A_1(Tw_1 - T) \quad (1)$$

$$h_2 = q/A_2(Ts - Tw_2) \quad (2)$$

h_1 = inside, or water side, film heat transfer coefficient BTU/hr-ft^2-$^{\circ}$F

h_2 = outside, or steam side, film heat transfer coefficient BTU/hr-ft^2-$^{\circ}$F

q = heat flux across tube wall, BTU/hr

A_1 = inside tube area, ft^2

A_2 = outside tube area, ft^2

Ts = condensing steam temperature, $^{\circ}$F

Tw_1 = inside tube wall temperature, $^{\circ}$F

Tw_2 = outside tube wall temperature, $^{\circ}$F

T = bulk water temperature inside tube, $^{\circ}$F

Since the tube wall temperatures are difficult to measure and are seldom known, the film coefficients must be determined indirectly by use of an overall heat transfer coefficient. The overall heat transfer coefficient based on inside tube area, for fluid flow in a round tube with steam condensing on the outside, is defined by:

$$q = U_1 A_1 \Delta Tm \quad (3)$$ where

U_1 = overall heat transfer coefficient, BTU/hr-ft^2-$^{\circ}$F

ΔT_m = log mean temperature difference causing heat transfer, ^oF. U_1 is generally a function of the geometry and dynamics of the heat transfer system. Since it is a conductivity U can be related to the individual heat transfer resistances:

$$U_1 = \frac{1}{R_1 + R_W + R_2} = \frac{1}{\frac{1}{h_1} + \frac{D_1 L}{D_m k} + \frac{D_1}{D_2 h_2}} \quad (4)$$

R_1 = water side resistance

R_W = tube wall resistance

R_2 = steam side resistance

D_1 = inside tube diameter, inches

D_2 = outside tube diameter, inches

D_m = average tube diameter, inches

L = tube wall thickness, inches

k_W = tube wall thermal conductivity, BTU-inch/hr-ft^2-^oF

U_1 is obtained experimentally in a heat exchanger by measuring the water inlet and outlet temperatures and flow rate, the condensing steam temperature outside the tube, and the inside tube diameter. These data are substituted, into the following equation which is obtained by equating the heat flow across the tube wall with the energy increase of the water flowing in the tube:

$$U_1 = \frac{60\,W}{A_1} \ln \frac{T_s - T_1}{T_s - T_2} = \frac{60\,W}{A_1} \ln \frac{T_s - T_1}{T_s - T_1 - \Delta T} \quad (5)$$

W = water flow rate, lbs/min

T_1 = inlet water temperature, ^oF

T_2 = outlet water temperature, ^oF

$\Delta T = T_2 - T_1$ = water temperature rise, ^oF

By use of the Wilson plot (Ref. 7) the individual coefficients are obtained from the overall coefficients. The Wilson method presupposes that the water side resistance R_1 is much larger than the steam side resistance R_2 and the wall resistance R_w. It is assumed that the sum R_2 plus R_w is constant. The water side resistance is an inverse function, F(V), of the water velocity V. For turbulent flow F(V) can be taken as $1/CV^{0.8}$ where C is a constant. From Equation 4:

$$1/U_1 = R_2 + R_w + R_1 = R_2 + R_w + 1/CV^{0.8} \qquad (6)$$

A plot of $1/U_1$ versus $1/CV^{0.8}$ should give a straight line when plotted on rectangular coordinates. The slope is equal to 1/C and the intercept I with the 1/U axis is equal to R_2 plus R_w. Thus:

$$1/U_1 = 1/h_1 + I \qquad (7)$$

$$h_1 = \frac{U}{I - UI}$$

An attempt will then be made to correlate h_1 empirically with the flow system variables. This correlation will probably be of the conventional form where the Nusselt number is related to the product of powers of other dimentionless groups such as Reynold's number, Prandtl number, viscosity ratio, and possibly the Weber number. Thus:

$$\frac{h_1 D_1}{K} = K\left(\frac{D_1 V \rho}{\mu}\right)^a \left(\frac{C\mu}{k}\right)^b \left(\frac{\mu}{\mu_1}\right)^c \left(\frac{\rho V^2 D_1}{\sigma g_c}\right)^d \qquad (8)$$

$\frac{h_1 D_1}{k}$ = Nusselt number

$\frac{D_1 V \rho}{\mu}$ = Reynold's number

$\frac{C\mu}{k}$ = Prandtl number

$\frac{\mu}{\mu_1}$ = Viscosity ratio

$\frac{\rho V^2 D_1}{\sigma g_c}$ = Weber number

K, a, b, c, and d are empirical constants

k = water thermal conductivity

ρ = water density

μ = bulk water viscosity

μ_1 = water viscosity at tube wall

C = water specific heat

σ = water surface tension

g_c = gravitational constant

The viscosity ratio may turn out to be unity for the purposes of this correlation because of small lateral water temperature variation across the tube. The Weber number is included because of possible surface tension effects due to non-wetting.

The final equation obtained will then be compared with correlations such as the Dittus-Boelter equation (Ref. 2) for wetted flow heat transfer. An attempt will then be made to explain any differences in terms of the heat transfer mechanisms involved.

DISCRIPTION OF APPARATUS

A diagram of the heat exchanger and the inlet water nozzle is shown in Figure I and the temperature measuring circuit in Figure II. The apparatus was a modification of that built by Tausch (Ref. 10) and Kessler and Regad (Ref. 8).

The apparatus was a one tube shell-and-tube heat exchanger with the Teflon tube mounted concentric with the six inch diameter, eight foot long outer shell which formed the steam chamber. The tube was supported on copper wires strung across the five inch diameter concentric insert which also acted as a steam diffuser.

The Teflon tube projected out the ends of the steam chamber through packing glands mounted on the headers. There was one foot of inlet calming length before the tube entered the steam chamber. The tube ends slipped over the inlet and outlet nozzles. The nozzles consisted of 3/4 inch crosses in which were mounted the water pipes, the thermocouple glands or thermometers, and an air vent. The glands or thermometers were mounted in the crosses opposite the pipes connecting the crosses with the Teflon tube. When the thermopile was used the lead wires extended into these pipes about a foot before they terminated at the thermocouple junctions. This arrangement provided for water flow around the lead wires to compensate for heat leak along the leads from outside the nozzles.

The outlet nozzle was the same as the inlet nozzle except that a 1/2 inch globe valve was situated immediately upstream of the thermocouple junctions. This acted as a mixer.

Two rotameters (0.35 to 3.55 gpm and 0 to 0.40 gpm) were mounted in parallel following the outlet nozzle. Flow control needle valves were mounted following the rotameters. The rotameters were calibrated and found to be accurate within the limits of the scales.

Two means of supplying water were provided. Water could be pumped from a hold tank to the inlet nozzle at a pressure of 24 psi. Alternately the city water pressure of 60 psi was used when the pump did not give the desired flow.

A six pair thermopile or a pair of thermometers were used to measure the water temperature rise in the tube. The thermocouple wire was Leeds and Northrup #20 high accuracy copper-constantan. Each end of the thermopile consisted of seven thermocouples. The extra thermocouple at the inlet nozzle was used to measure the inlet water temperature. A thermocouple was also located in the steam chamber.

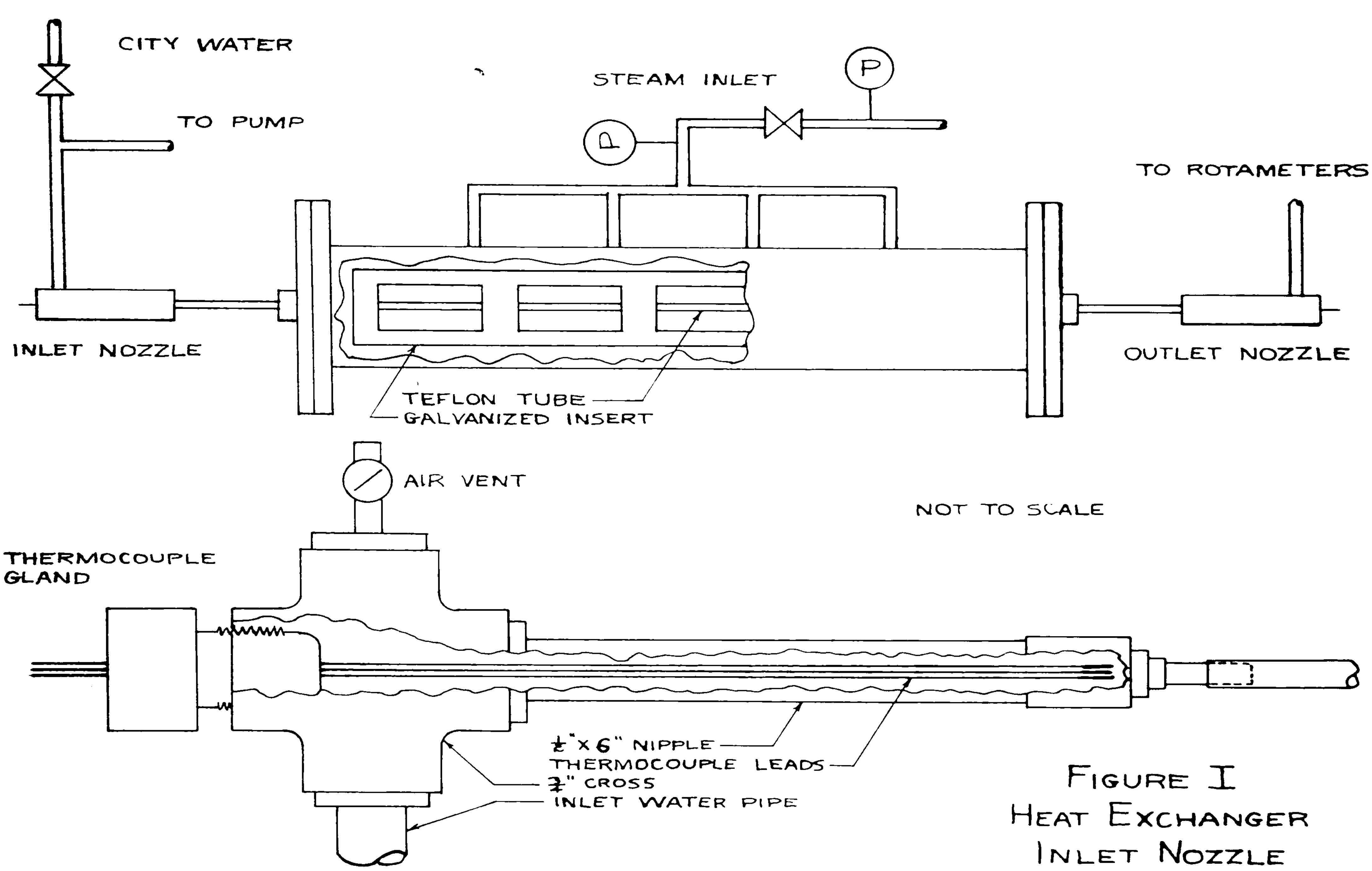
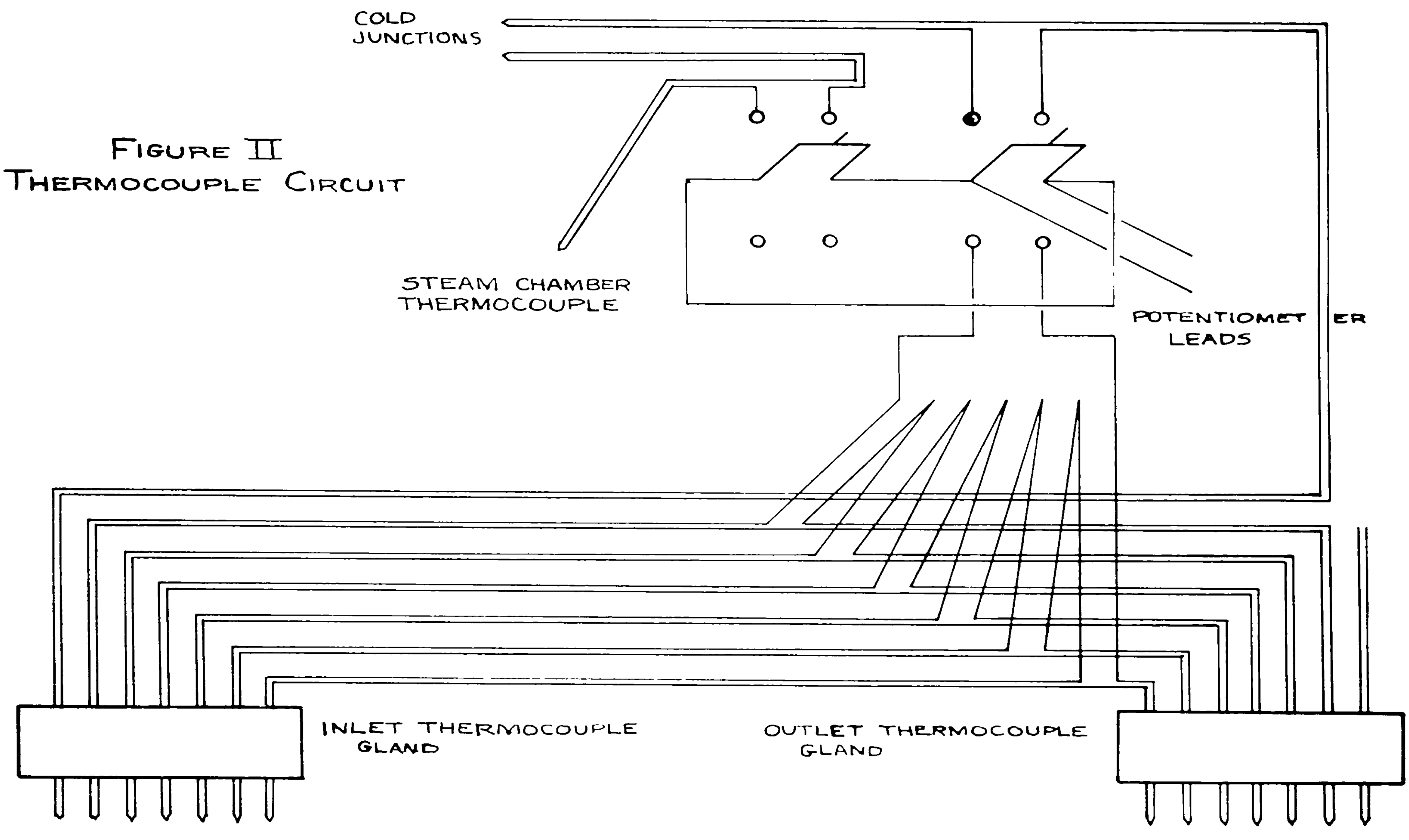


FIGURE I
HEAT EXCHANGER
INLET NOZZLE

FIGURE II
THERMOCOUPLE CIRCUIT



EXPERIMENTAL PROCEDURE

The length of a Teflon tube and the volume of water needed to fill the tube were measured. The average inside diameter of the tube was calculated from these data.

The Teflon tube was then cleaned with soap solution followed by a dilute acid rinse. It was then mounted in the steam chamber and the headers bolted on. The packing glands were stuffed and tightened. The projecting tube ends were forced over the inlet and outlet nozzle tips. The tube was kept relatively straight by allowing the nozzles to exert a slight tension on the tube ends. The thermopile was screwed into the nozzles and connected to the circuit. Thermometers were used in place of the thermopile for several runs.

Water flow through the tube was obtained by pumping water from the hold tank or by using water directly from the city water mains. Water was not recycled.

Steam was run into the steam chamber and allowed to flush air out for ten minutes. After this time the steam pressure was adjusted to the desired value. A small steam purge was maintained to remove air.

The water flow rate was set at the desired value with the throttling valve downstream of the tube. Voltages of the inlet temperature thermocouple, the thermopile, the steam chamber thermocouple were recorded along with the steam pressure. These readings were repeated until constant values, particularly of the thermopile voltage, were obtained. When thermometers were used to measure inlet and outlet water temperatures these readings were recorded. The flow rate was then changed and the process repeated.

Representative values of each of the data for each flow rate were used to calculate the overall coefficients. Runs were made on 1/4, 5/16, and 3/8 inch nominal I. D. tubes of 30 mil wall thickness.

RESULTS

Following are the experimental data, including water flow rate, water temperature rise, inlet water temperature, and steam temperature, and the corresponding calculated values of Reynolds number and overall heat transfer coefficient, for the 3/8 inch nominal I. D. Teflon tube.

TABLE I

Data of September 8, 1959. Low water pressure was used except for the runs marked "H" where high pressure was used.

FLOW RATE lbs/MIN	TEMP. RISE °F	INLET TEMP °F	STEAM TEMP °F	REYNOLDS NUMBER	HEAT TRANS COEFF. BTU/HR-FT^2°F
26.6	4.18	68.3	212	28200	58.1
20.7	5.19			21900	56.3
14.8	6.90			15700	53.8
8.87	10.86			9400	51.6
2.96	28.50			3140	48.4
5.91	15.53			6280	50.0
11.8	8.31			12500	52.0
17.7	5.81			18700	54.0
23.7	4.54			25100	56.2
26.6	4.12			28100	57.2
20.7	5.10			21900	55.3
29.6 H	3.74			31400	57.7
20.7 H	5.07			21900	55.0
8.87 H	10.74			9400	51.0

TABLE II

Data of September 16, 1959. Low water pressure was used.

FLOW RATE lbs/MIN	TEMP RISE oF	INLET TEMP oF	STEAM TEMP oF	REYNOLDS NUMBER	HEAT TRANS COEFF BTU/HR-FT^2 oF
14.8	6.32	67.4	212	15700	48.9
17.7	5.36	67.4		18700	49.4
20.7	4.59	67.4		21900	49.4
23.7	4.03	67.4		25100	49.6
26.6	3.58	67.4		28200	49.3
8.87	10.18	67.4		9400	47.9
5.91	14.88	68.0		6280	47.6
2.92	26.55	68.3		3100	44.1
11.8	7.85	68.3		12500	49.0

TABLE III

Data of September 17, 1959. Low water pressure was used.

FLOW RATE lbs/MIN	TEMP RISE oF	INLET TEMP oF	STEAM TEMP oF	REYNOLDS NUMBER	HEAT TRANS COEFF BTU/HR-FT^2-oF
5.91	14.91	68.3	212	6280	47.9
11.8	7.54	68.3		12500	47.0
17.7	5.06	67.6		18700	46.5
23.7	3.80	67.5		25100	46.7
26.6	3.40	67.1		28200	46.6
20.7	4.41	67.1		21900	47.3
14.8	6.15	67.6		15700	47.6
8.87	10.10	68.0		9400	47.7
2.96	28.01	68.3		3140	47.5

TABLE IV

Data of September 17, 1959. High water pressure was used.

FLOW RATE lbs/MIN	TEMP RISE °F	INLET TEMP °F	STEAM TEMP °F	REYNOLDS NUMBER	HEAT TRANS COEFF BTU/HR-FT^2-°F
17.7	4.85	68.3	212	18700	44.9
11.8	7.39	67.9		12500	46.0
5.91	14.66	68.3		6280	47.0
23.7	3.55	67.7		25100	43.6
29.6	2.80	67.5		31400	42.9
26.6	3.17	67.4		28100	43.6
20.7	4.18	67.3		21900	44.8
14.8	5.94	67.3		15700	45.8
8.87	9.93	67.6		9400	46.7
2.96	27.91	68.0		3140	47.2

Following are the experimental data including water flow rates inlet and outlet temperatures, and steam temperature, and the corresponding calculated values of water temperature rise, Reynolds number, and overall heat transfer coefficient, for the 3/8 inch nominal I. D. Teflon tube. Thermometers were used for these runs.

TABLE V

Data of September 22, 1959. High water pressure was used.

FLOW RATE lbs/MIN	INLET TEMP °F	OUTLET TEMP °F	STEAM TEMP °F	TEMP RISE °F	REYNOLDS NUMBER	HEAT TRANS COEFF BTU/HR-FT^2-°F
26.6	67.70	71.39	212	3.69	28200	50.9
20.7	67.43	72.18		4.75	21900	51.2
14.8	67.43	73.97		6.54	15700	50.7
8.87	67.53	78.00		10.47	9400	49.3
2.96	68.10	95.80		27.70	3140	46.8
5.91	67.95	83.14		15.19	6280	48.7
11.8	67.81	75.81		8.00	12500	49.8
17.7	67.88	73.35		5.47	18700	50.7
23.7	67.75	71.90		4.15	25100	51.1
29.6	67.70	70.00		3.30	31400	50.6

TABLE VI

Data of September 22, 1959. Low water pressure was used.

FLOW RATE lbs/MIN	INLET TEMP of	OUTLET TEMP of	STEAM TEMP of	TEMP RISE of	REYNOLDS NUMBER	HEAT TRANS COEFF BTU/HR-FT^2-^{o}F
5.91	69.48	84.38	212	14.90	6280	48.3
11.8	69.04	76.94		7.90	12500	49.6
17.7	68.90	74.30		5.40	18700	50.3
23.7	68.81	72.90		4.09	25100	50.8
28.5	68.79	72.14		3.35	30200	50.0
20.7	68.90	73.55		4.65	21900	50.6
14.8	69.07	75.43		6.36	15700	49.8

TABLE VII

Data of September 23, 1959. Low water pressure was used.

FLOW RATE lbs/MIN	INLET TEMP of	OUTLET TEMP of	STEAM TEMP of	TEMP RISE of	REYNOLDS NUMBER	HEAT TRANS COEFF BTU/HR-FT^2-^{o}F
29.6	67.46	70.71	212	3.25	31400	49.8
23.7	67.59	71.72		4.13	25100	50.8
17.7	67.73	73.19		5.46	18700	50.5
11.8	67.92	75.89		7.97	12500	49.7
5.91	68.21	83.30		15.09	6280	48.4
2.96	68.43	96.20		27.77	3140	47.0
8.87	68.03	78.51		10.48	9400	49.6
14.8	67.82	74.30		6.48	15700	50.4
20.7	67.70	72.39		4.69	21900	50.6
26.6	67.60	71.25		3.65	28100	50.3

Following are the experimental data, including water flow rate, water temperature rise, inlet water temperature, and steam temperature, and the corresponding calculated values of Reynolds number and overall heat transfer coefficient for the 5/16 inch nominal I. D. Teflon tube.

TABLE VIII

Data of September 18, 1959. Low water pressure was used.

FLOW RATE lbs/MIN	TEMP RISE of	INLET TEMP of	STEAM TEMP of	REYNOLDS NUMBER	HEAT TRANS COEFF BTU/HR-FT^2-^{o}F
23.5	4.18	66.7	213	31000	61.9
17.7	5.35	66.9		23300	60.0
11.8	7.65	66.9		15600	57.7
5.91	14.47	66.9		7790	56.0
2.96	27.52	66.5		3900	56.2
8.87	9.90	66.3		11700	56.4
14.8	6.15	66.5		19500	57.7
20.7	4.67	66.5		27300	60.9
23.7	4.18	66.5		31000	62.4

TABLE IX

Data of September 18, 1959. Low water pressure was used.

FLOW RATE lbs/MIN	TEMP RISE of	INLET TEMP of	STEAM TEMP of	REYNOLDS NUMBER	HEAT TRANS COEFF BTU/HR-FT^2-^{o}F
5.91	14.09	67.2	213	7790	54.6
11.8	7.33	67.0		15500	55.3
17.7	5.02	66.8		23300	56.3
23.7	3.92	66.5		31200	58.4
29.6	3.27	66.2		39000	60.7
26.6	3.60	66.2		35000	59.9
20.7	4.48	66.2		27300	58.3
14.8	6.06	66.5		19500	56.9
8.87	9.79	66.8		11700	55.9
[illegible].96	27.39	67.5		3900	56.1

TABLE X

Data of September 23, 1959. Low water pressure was used. Water temperatures were measured with thermometers.

FLOW RATE lbs/MIN	INLET TEMP oF	OUTLET TEMP oF	STEAM TEMP oF	TEMP RISE oF	REYNOLDS NUMBER	HEAT TRANS COEFF BTU/HR-FT^2-°F
23.2	69.12	72.80	212	3.68		
17.7	69.30	74.10		4.80		
11.8	69.88	76.90		7.02	15500	54.4
11.8	58.73	75.98		7.25	15500	55.7
5.91	69.61	83.32		13.71	7790	54.4

TABLE XI

Data of September 24, 1959. Low water pressure was used. Water temperatures were measured with thermometers.

FLOW RATE lbs/MIN	INLET TEMP of	OUTLET TEMP of	STEAM TEMP of	TEMP RISE of	REYNOLDS NUMBER	HEAT TRANS COEFF BTU/HR-FT ^{2}-^{o}F
2.96	69.07	95.84	212	26.77	3900	55.8
8.87	68.70	78.19		[illegible]	11700	55.2
14.8	68.47	74.30		5.83	19500	55.7
20.7	68.63	72.80		4.17	27300	55.6
25.1	68.50	71.90		3.40	33000	54.7
23.7	68.53	72.19		3.66	31200	55.7
17.7	68.61	73.50		4.89	23300	55.8
11.8	68.20	75.35		7.15	15500	54.7
5.91	68.44	82.36		13.92	7790	54.8

TABLE XII

Data of September 24, 1959. High water pressure was used. Water temperatures were measured with thermometers.

5.91	68.25	82.30	212	14.05	7790	55.2
11.8	68.10	75.40		7.30	15500	55.8
17.7	68.36	73.24		4.88	23300	55.7
23.7	68.15	71.80		3.65	31200	55.3
29.6	68.07	70.90		2.83	39000	53.5
26.6	68.21	71.42		3.21	35000	54.5
20.7	68.45	72.65		4.20	27300	55.8
14.8	68.41	74.27		5.86	19500	56.0
8.87	68.34	77.97		9.63	11700	55.9
2.96	68.50	95.32		26.82	3900	55.7

Following are the experimental data, including water flow rate, water temperature rise, inlet water temperature, and steam temperature, and the corresponding calculated values of Reynolds number and overall heat transfer coefficient for the 1/4 inch nominal I. D. Teflon tube.

TABLE XIII

Data of September 14, 1959. High water pressure was used.

FLOW RATE lbs/MIN	TEMP RISE of	INLET TEMP of	STEAM TEMP of	REYNOLDS NUMBER	HEAT TRANS COEFF BTU/HR-FT^2-^{o}F
2.96	24.33	68.8	212	4930	64.3
8.87	7.83	68.3		14800	58.0
14.8	4.89	68.3		24700	59.7
20.7	3.70	68.1		34500	62.9
17.7	4.24	68.1		29500	61.8
11.8	6.03	68.3		19700	59.0
5.91	11.59	68.3		9880	58.0

TABLE XIV

Data of September 14, 1959. High water pressure was used.

FLOW RATE lbs/MIN	TEMP RISE oF	INLET TEMP oF	STEAM TEMP oF	REYNOLDS NUMBER	HEAT TRANS COEFF BTU/HR-FT^2-^{o}F
2.96	22.71	68.3	212	4930	59.4
5.91	11.17	68.3		9880	55.8
3.33	18.63	68.3		15550	53.9
8.87	7.46	68.3		14800	55.2
11.8	5.85	67.2		19700	56.8
14.8	4.84	67.2		24700	58.7
17.7	4.33	67.2		29500	62.6
23.7	3.48	67.1		39600	67.2
20.7	4.23	66.8		34500	71.4
8.87	8.32	66.9		14800	61.1
20.7	4.11	66.5		34500	69.2
23.7	3.75	66.9		39600	72.3

TABLE XV

Data of September 15, 1959. High water pressure was used except for the runs marked "L", where low pressure was used.

20.7	4.32	67.1	212	34500	73.0
11.8	5.74	68.1		19700	60.0
5.91	11.18	68.0		9880	55.7
5.91 L	11.01	69.2		9880	55.3
2.96 L	22.24	68.7		4930	58.2
2.08 L	27.80	68.7		3460	52.3
3.33 L	18.50	68.3		5550	53.6
13.7 L	4.58	68.7		22900	51.9
8.87	7.19	69.4		14800	53.5
17.7	3.23	67.5		29500	46.8

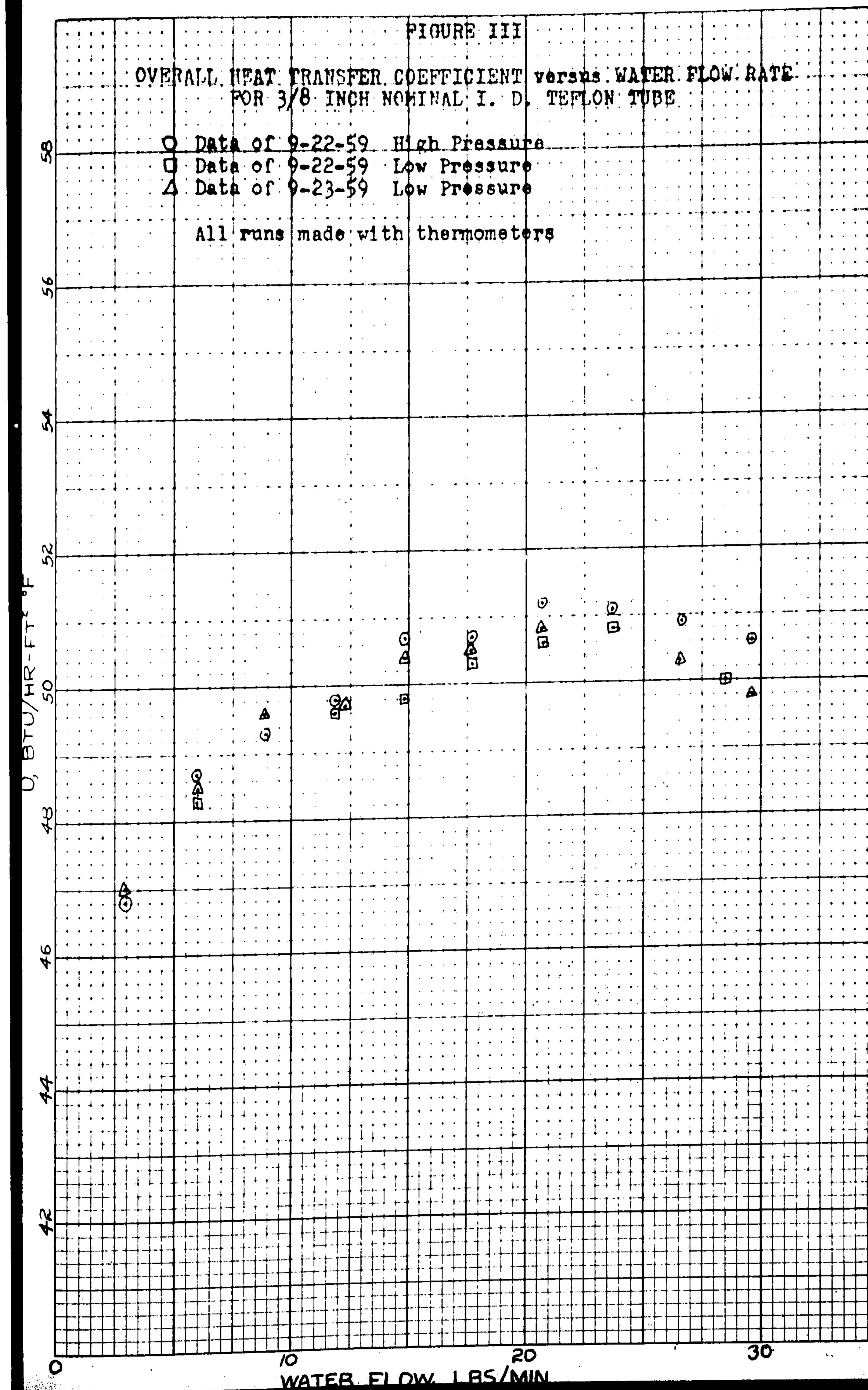


FIGURE III

OVERALL HEAT TRANSFER COEFFICIENT versus WATER FLOW RATE FOR 3/8 INCH NOMINAL I. D. TEFLON TUBE

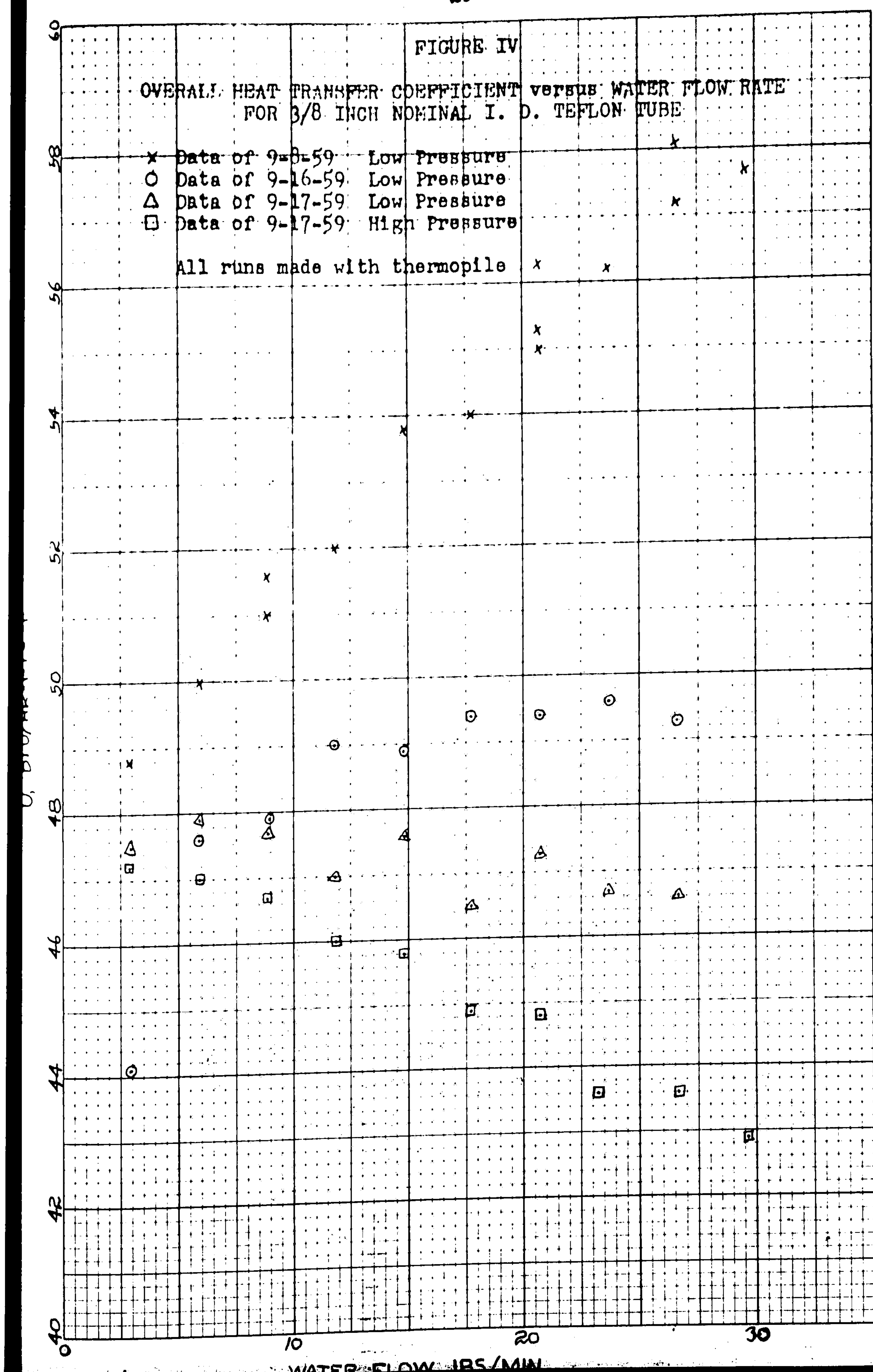
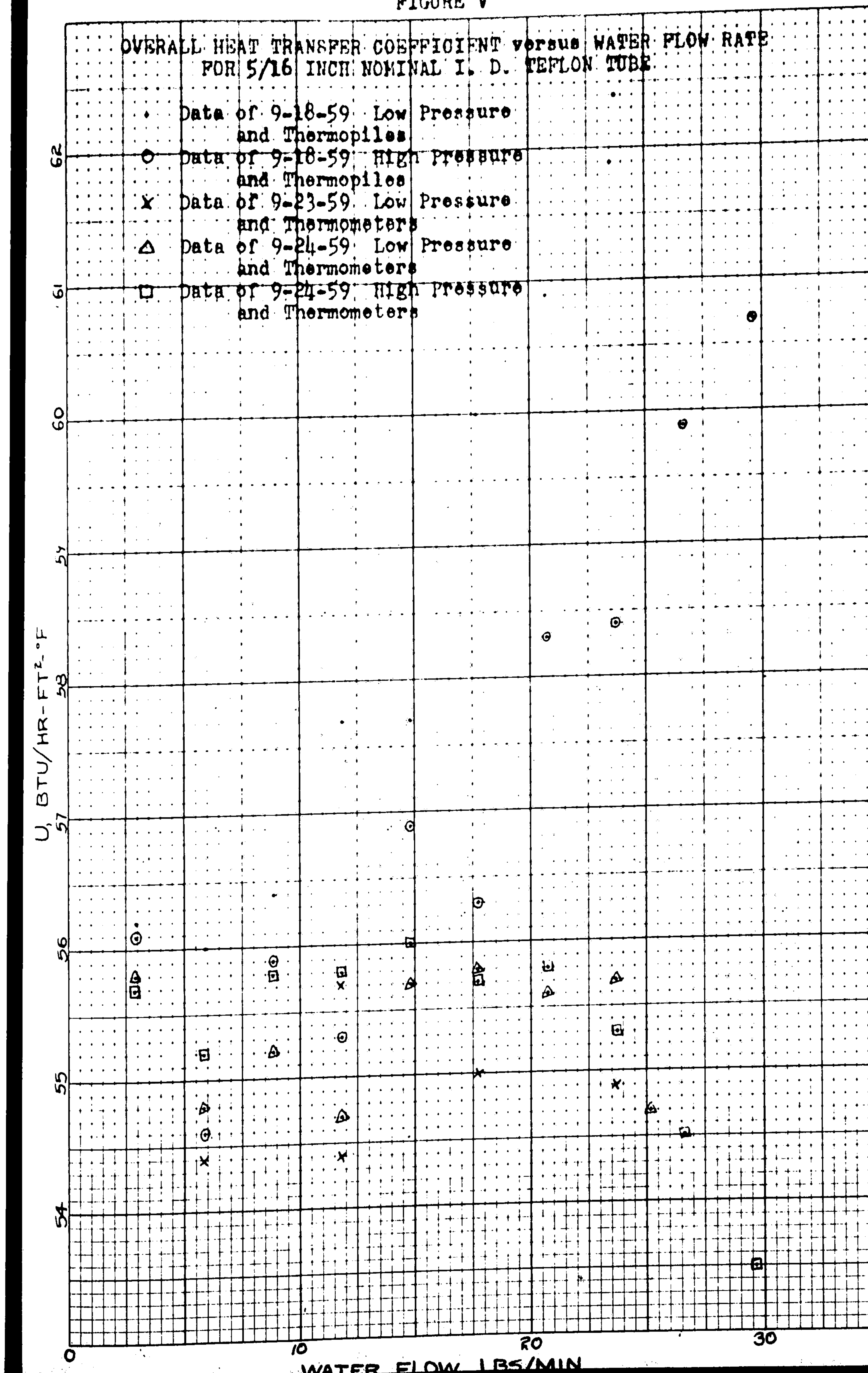


FIGURE IV
OVERALL HEAT TRANSFER COEFFICIENT versus WATER FLOW RATE
FOR 3/8 INCH NOMINAL I. D. TEFLON TUBE
Data of 9-8-59 Low Pressure
Data of 9-16-59 Low Pressure
Data of 9-17-59 Low Pressure
Data of 9-17-59 High Pressure
All runs made with thermopile
60
58
56
54
52
50
48
46
44
42
40
0
10
20
30
WATER FLOW LBS/MIN

FIGURE V



OVERALL HEAT TRANSFER COEFFICIENT versus WATER FLOW RATE
FOR 5/16 INCH NOMINAL I. D. TEFLON TUBE
Data of 9-18-59 Low Pressure and Thermopiles
Data of 9-18-59 High Pressure and Thermopiles
Data of 9-23-59 Low Pressure and Thermometers
Data of 9-24-59 Low Pressure and Thermometers
Data of 9-24-59 High Pressure and Thermometers
U, BTU/HR-FT²-°F
54
55
56
57
58
59
60
61
62
0
10
20
30
WATER FLOW LBS/MIN

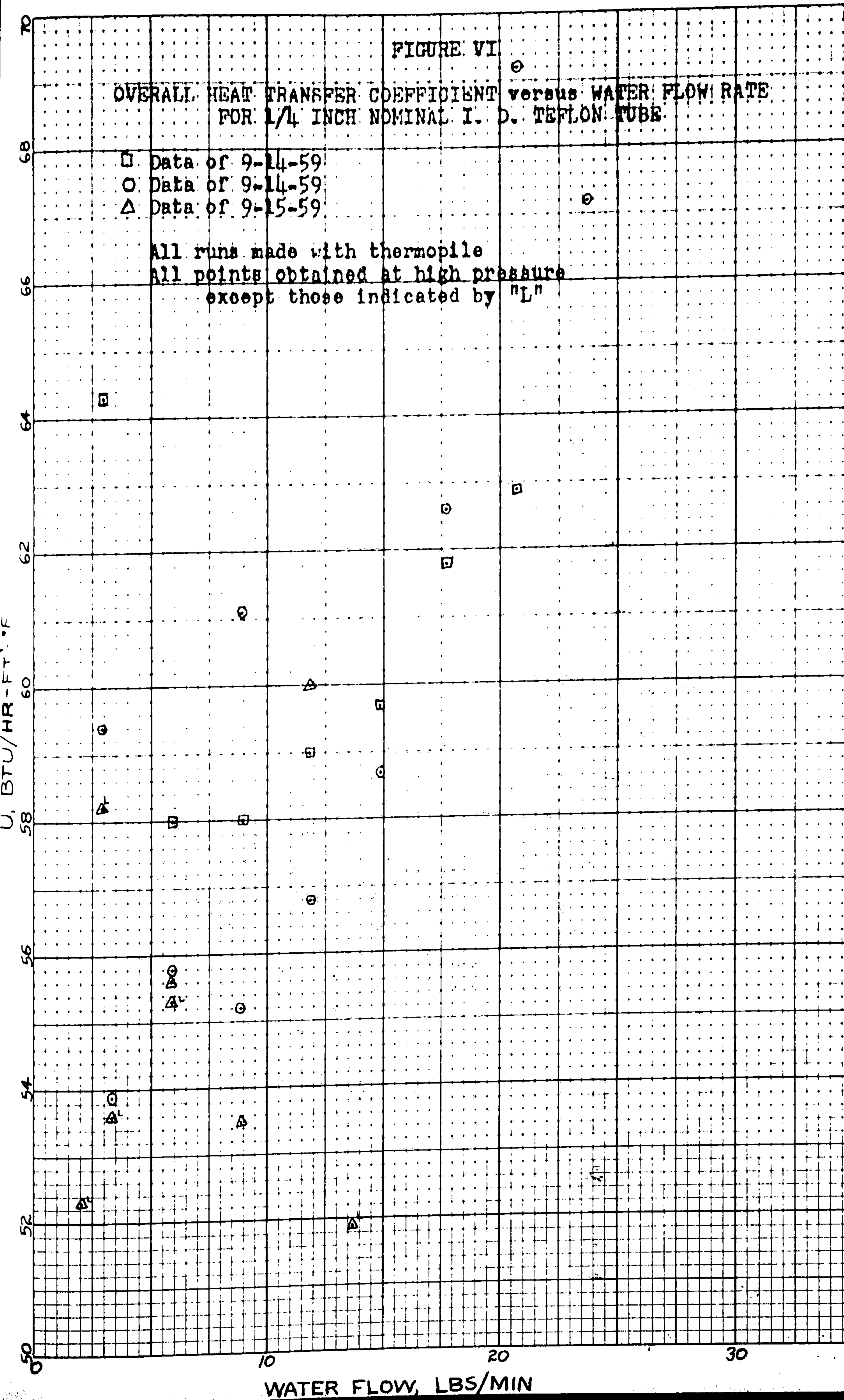
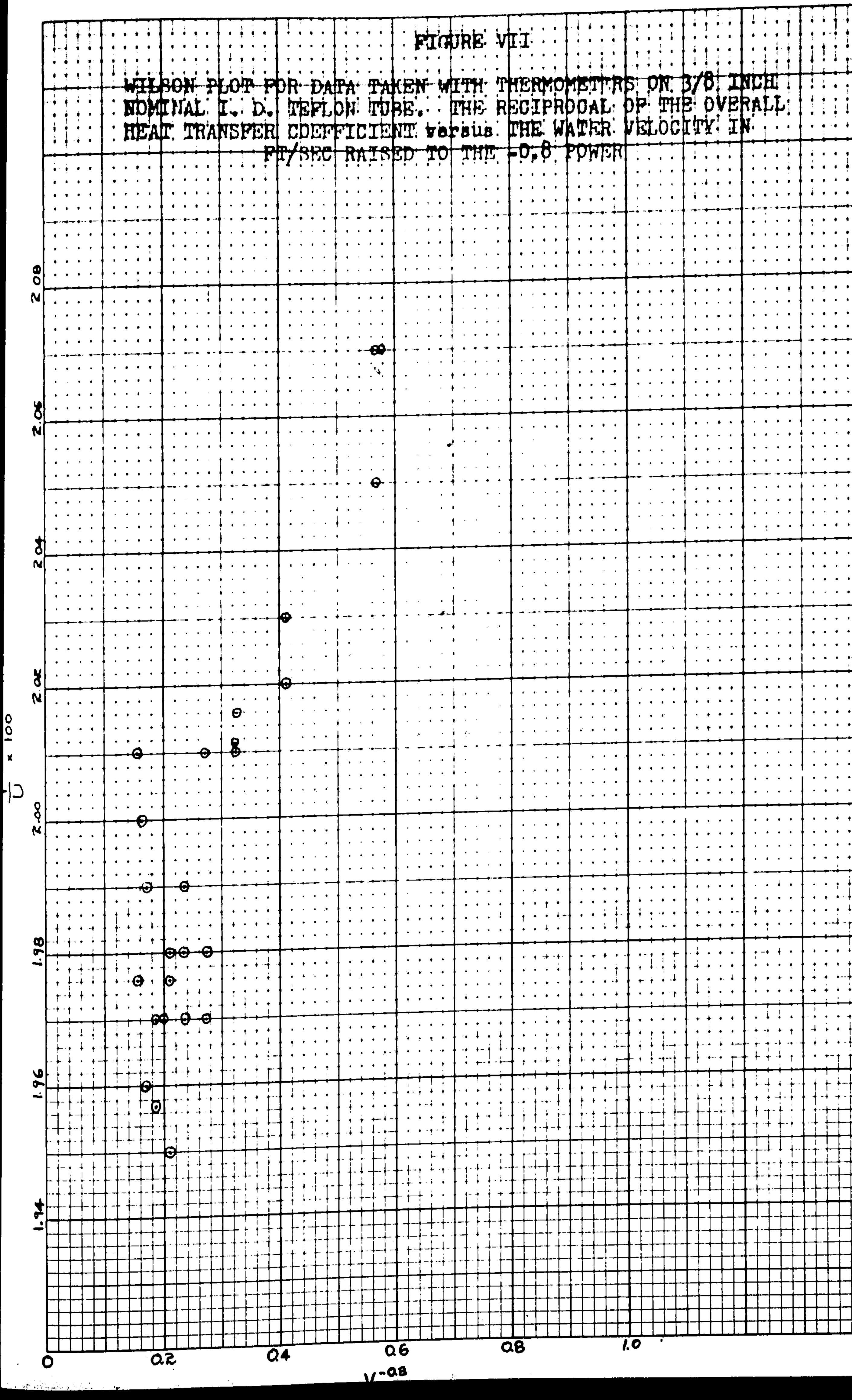


FIGURE VI

OVERALL HEAT TRANSFER COEFFICIENT versus WATER FLOW RATE
FOR 1/4 INCH NOMINAL I. D. TEFLON TUBE

FIGURE VII

WILSON PLOT FOR DATA TAKEN WITH THERMOMETERS ON 3/8 INCH NOMINAL I. D. TEFLON TUBE. THE RECIPROCAL OF THE OVERALL HEAT TRANSFER COEFFICIENT versus THE WATER VELOCITY IN FT/SEC RAISED TO THE -0.8 POWER



DISCUSSION OF RESULTS

The experimental data and the calculated overall heat transfer coefficients obtained in this investigation for the 3/8, 5/16, and 1/4 inch Teflon tubes are shown in Tables I to XV. Graphs of the coefficients plotted against water flow rates are shown in Figures III to VI. Figure VII is a Wilson plot for the 3/8 inch tube.

The results leave much to be desired in the way of consistency and reproducibility. Three factors seem to contribute to this.

(1) The calculated values of the overall heat transfer coefficients are inherently very sensitive to error in the measured water temperature rise. Differentiating Equation 5 with respect to ΔT we obtain:

$$\frac{\partial U_1}{\partial \Delta T} = \frac{60\ W}{A_1\ (T_s - T_2)} \qquad \text{or approximately} \qquad (9)$$

$$\Delta U_1 = \frac{60\ W}{A_1\ (T_s - T_2)} \times \Delta(\Delta T) \qquad (10)$$

It can be seen that for a given error in ΔT the error in U_1 is directly proportional to W. This helps explain the greater variation of values at high flow rates than at low flow rates.

For the 3/8 inch tube Equation 9 becomes:

$$U_1 = \frac{60\ W}{0.811 \times (212 - 75)} \times \Delta(\Delta T) = 0.54\ W\ \Delta(\Delta T) \qquad (11)$$

When W = 25 lbs/min $U_1 = 13.5\ \Delta(\Delta T)$ and an error of 0.1 ^oF in measuring ΔT leads to an error in U_1 of 1.4 BTU/hr-ft^2-^oF or about 2.7%. This error in U_1 goes up proportionally with the error in ΔT.

(2) The low modulus of elasticity of Teflon and its tendency to creep may lead to changes in the tube diameters. Teflon is an elastic material subject to creep under stress. When a deforming load is removed from Teflon there is some recovery even with no temperature change. The amount of recovery increases with increasing temperature applied after removing the load and complete recovery is effected at the transition temperature of 620°F (Ref. 11). Renfrew and Lewis (Ref. 9) report:

> It is not possible to establish a modulus of elasticity for Teflon in the normal sense, but at a no-load cross-head speed of 0.05 inch per minute, values around 1700 psi have been recorded at deformations of 0.1% under compression.

The experimental runs were made with pump pressure or city water pressure. The pump produced a tube inlet pressure of 24 psi and the city water pressure was about 60 psi. At higher flow rates, particularly with the smaller tubes, the static pressure in the tube varied as much as 50 psi over the length of the steam chamber. At low rates the pressure was high all along the tube. In other words, in the course of a run the static pressure at a point in a tube might vary as much as 50 psi.

The change in diameter of a thin walled tube with change in internal pressure is given by:

$$\Delta D = \frac{\Delta P D^2}{2tY} \quad \text{where} \qquad (12)$$

D = tube diameter, inches

ΔD = change in tube diameter, inches

ΔP = change in pressure inside tube, psi t = tube wall thickness, inches

Y = modulus of elasticity, psi

For the 5/16 inch tube the change in diameter at a point with a 50 psi pressure change, using the above value for the modulus of elasticity, is:

$$D = \frac{50 \times (0.308)^2}{2 \times 0.03 \times 1700} = 0.046 \text{ inches or } 46 \text{ mils}$$

This is a 15% change in tube diameter.

Expressing Equation 5 in terms of D_1 and differentiating with respect to D_1 gives:

$$\frac{\partial U_1}{\partial D_1} = -\frac{2.33_2 W}{D_1} \ln \frac{T_s - T_1}{T_s - T_1 - \Delta T} \qquad (13)$$

Substituting data for the 5/16 inch tube at a flow rate of 26.6 lbs/min yields:

$$\frac{\partial U_1}{\partial D_1} = -\frac{2.33 \times 26.6}{(0.0256)^2} \ln \frac{212 - 66.2}{212 - 66.2 - 3.60} = -1010 \qquad (14)$$

or approximately $\Delta U_1 = 1010 \, \Delta D_1$

A one mil change in D_1 leads to a variation in U_1 of 1.0 or about 2%. Changes in D_1 of the order which could be produced by the pressure changes which occured in the tubes is sufficient to account for the unreproducibility of the overall coefficients. The high temperature gradients across the tube walls could also be expected to increase the distortion caused by pressure changes.

(3) The individual thermocouples of a thermopile must be perfectly insulated from each other and from ground in order for

the thermopile to be accurate. Great care was taken to insulate the thermocouples and their leads. However the electrical insulation could not be made too heavy or there would not have been enough heat transfer between the water and the thermocouple wires to compensate for the heat conduction along the thermocouple lead wires. Several electrical leaks were found in the first thermopiles that were used but it is believed that the ones used for taking the data in this report were in good condition. However the possibility remains that ill-functioning thermopiles contributed to the irratic results. The data taken using thermometers is more consistent, particularly in the case of the 3/8 inch tube.

(4) It was impossible to keep the Teflon tubes straight in the heat exchanger. The curving flow paths may have caused some slight variation among runs.

CONCLUSIONS

The use of Teflon tubes for measuring heat transfer coefficients is unsatisfactory due to the elasticity of the tubes and to the high wall heat transfer resistance. An investigation of this type should be carried out in Teflon coated steel or copper pipes. Methods are available (Ref. 11) for coating Teflon on metal by dipping the metal into an aqueous dispersion of Teflon followed by baking. It is also possible to buy commerical Teflon lined pipes. The use of rigid pipes would also eliminate the crooked flow paths obtained with the flexible tubes.

Thermometers should be used instead of home-made thermopiles for the water temperature rise measurements. The thermopiles are too prone to short-circuit or to have heat leaks through the leads.

APPENDIX

SAMPLE CALCULATIONS

The following is a sample calculation based on the first data point for the 3/8 inch tube given on page 32. The tube area calculation is for the 3/8 inch tube.

Calculation of inside heat transfer area

Length of tube used for volume measurement = 10.58 ft

Volume of water used to fill tube (average of 3 trials) = 233 cc

$$D_1 = 0.00670 \sqrt{V/L} = 0.00670 \sqrt{233/10.58} = 0.0315 \text{ ft}^2$$

Length of tube in heat exchanger steam chamber = 8.21 ft

$$A_1 = \pi D_1 L = \pi \times 0.0315 \times 8.21 = 0.811 \text{ ft}^2$$

Calculation of overall heat transfer coefficient

Water flow rate	26.6 lbs/min
Inlet thermocouple voltage	0.80 millivolts
Thermopile voltage	0.563 millivolts
Steam temperature	212 °F

Thermopile voltage per pair of thermocouples = 0.563/6 = 0.0938 millivolts

$$\Delta T = 0.0938 \text{ mv} \times 44.58 \text{ °F/mv} = 4.18 \text{ °F}$$

From copper-constantan thermocouple conversion table

$$T_1 = 68.3 \text{ °F}$$

$$U_1 = \frac{60W}{A_1} \ln \frac{T_s - T_1}{T_s - T_2} = \frac{60W}{A_1} \ln \frac{T_s - T_1}{T_s - T_1 - \Delta T}$$

$$= \frac{60 \times 26.6}{0.811} \ln \frac{212 - 68.3}{212 - 68.3 - 4.18} = 58.1 \text{ BTU/hr-ft}^2\text{-°F}$$

Calculation of Reynold's number

$$Re = \frac{D_1 V \rho}{\mu}$$

Average water temperature = 70 °F

ρ = 62.28 lbs/ft^3 at 70 °F

μ = 0.975 centipoises at 70 °F

= 0.975 x 1488 lb mass/ft-sec

$$V = \frac{26.6 \text{ lbs/min}}{62.28 \text{ lbs/ft}^3 \times 60 \text{ sec/min} \times 0.000778 \text{ ft}^2} = 9.1 \text{ ft/sec}$$

$$Re = \frac{0.0315 \times 9.1 \times 62.28 \times 1488}{0.975} = 27{,}400$$

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